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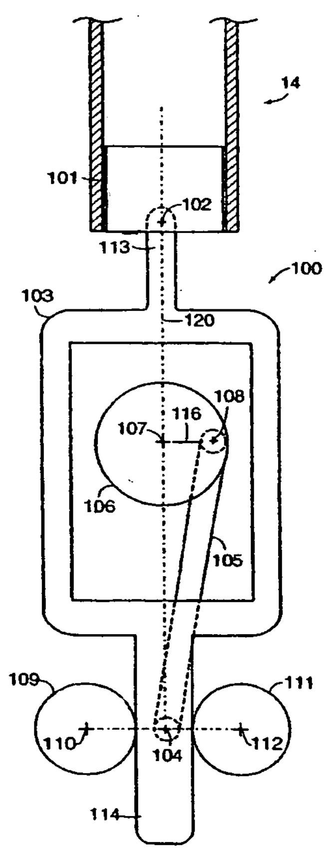
(54) Title: FOLDED GUIDE LINK STIRLING ENGINE



(57) Abstract: A folded linkage (100) for coupling a crankshaft (106) and a piston (101) undergoing reciprocating linear motion along a longitudinal axis. The folded linkage (100) has a guide link (103) with a first end coupled to the piston (101). A connecting rod (105) couples the distal end of the guide link (103) to the crankshaft (106) which rotates about an axis that is orthogonal to the longitudinal axis (120) of piston motion and located between the proximal end and the distal end of the guide link (103). A guide link guide assembly supports lateral loads on the guide link (103) at its distal end. The folded linkage (100) may be applied to couple the compression piston (311) and displacer piston (301) of a Stirling cycle machine to a common crankshaft (106).

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Folded Guide Link Stirling Engine

Technical Field

The present invention pertains to improvements to an engine and more particularly to improvements relating to mechanical components of a Stirling cycle heat engine or refrigerator which contribute to increased engine operating efficiency and lifetime, and to reduced size, complexity and cost.

Background of the Invention

Stirling cycle machines, including engines and refrigerators, have a long technological heritage, described in detail in Walker, *Stirling Engines*, Oxford University Press (1980), herein incorporated by reference. The principle underlying the Stirling cycle engine is the mechanical realization of the Stirling thermodynamic cycle: isovolumetric heating of a gas within a cylinder, isothermal expansion of the gas (during which work is performed by driving a piston), isovolumetric cooling, and isothermal compression. The Stirling cycle refrigerator is also the mechanical realization of a thermodynamic cycle which approximates the ideal Stirling thermodynamic cycle. In an ideal Stirling thermodynamic cycle, the working fluid undergoes successive cycles of isovolumetric heating, isothermal expansion, isovolumetric cooling and isothermal compression. Practical realizations of the cycle, wherein the stages are neither isovolumetric nor isothermal, are within the scope of the present invention and may be referred to within the present description in the language of the ideal case without limitation of the scope of the invention as claimed.

Various aspects of the present invention apply to both Stirling cycle engines and

Stirling cycle refrigerators, which are referred to collectively as Stirling cycle machines in the present description and in any appended claims. The principle of operation of a Stirling cycle engine configured in an 'alpha' configuration and employing a first "compression" piston and a second "expansion" piston is described at length in pending U.S. application 09/115,383, filed July 14, 1998, which is incorporated herein by reference.

The principle of operation of a Stirling engine is readily described with reference to FIGS. 1a-1e, wherein identical numerals are used to identify the same or similar parts. Many mechanical layouts of Stirling cycle machines are known in the art, and the particular Stirling engine designated generally by numeral **10** is shown merely for illustrative purposes. In FIGS. 1a to 1d, piston **12** and a displacer **14** move in phased reciprocating motion within cylinders **16** which, in some embodiments of the Stirling engine, may be a single cylinder. Typically, a displacer **14** does not have a seal. However, a displacer **14** with a seal (commonly known as an expansion piston) may be used. Both a displacer without a seal or an expansion piston will work in a Stirling engine in an "expansion" cylinder. A working fluid contained within cylinders **16** is constrained by seals from escaping around piston **12** and displacer **14**. The working fluid is chosen for its thermodynamic properties, as discussed in the description below, and is typically helium at a pressure of several atmospheres. The position of displacer **14** governs whether the working fluid is in contact with hot interface **18** or cold interface **20**, corresponding, respectively, to the interfaces at which heat is supplied to and extracted from the working fluid. The supply and extraction of heat is discussed in further detail below. The volume of working fluid governed by the position of the piston **12** is referred to as compression space **22**.

During the first phase of the engine cycle, the starting condition of which is depicted in FIG. 1a, piston **12** compresses the fluid in compression space **22**. The compression occurs

at a substantially constant temperature because heat is extracted from the fluid to the ambient environment. In practice, a cooler (not shown) is provided. The condition of engine **10** after compression is depicted in FIG. 1b. During the second phase of the cycle, displacer **14** moves in the direction of cold interface **20**, with the working fluid displaced from the region of cold interface **20** to the region of hot interface **18**. This phase may be referred to as the transfer phase. At the end of the transfer phase, the fluid is at a higher pressure since the working fluid has been heated at constant volume. The increased pressure is depicted symbolically in FIG. 1c by the reading of pressure gauge **24**.

During the third phase (the expansion stroke) of the engine cycle, the volume of compression space **22** increases as heat is drawn in from outside engine **10**, thereby converting heat to work. In practice, heat is provided to the fluid by means of a heater (not shown). At the end of the expansion phase, compression space **22** is full of cold fluid, as depicted in FIG. 1d. During the fourth phase of the engine cycle, fluid is transferred from the region of hot interface **18** to the region of cold interface **20** by motion of displacer **14** in the opposing sense. At the end of this second transfer phase, the fluid fills compression space **22** and cold interface **20**, as depicted in FIG. 1a, and is ready for a repetition of the compression phase. The Stirling cycle is depicted in a P-V (pressure-volume) diagram as shown in FIG. 1e.

Additionally, on passing from the region of hot interface **18** to the region of cold interface **20**, the fluid may pass through a regenerator (not shown). The regenerator may be a matrix of material having a large ratio of surface area to volume which serves to absorb heat from the fluid when it enters hot from the region of hot interface **18** and to heat the fluid when it passes from the region of cold interface **20**.

The principle of operation of a Stirling cycle refrigerator can also be described with reference to FIGS. 1a-1e, wherein identical numerals are used to identify the same or similar parts. The differences between the engine described above and a Stirling machine employed as a refrigerator are that compression volume **22** is typically in thermal communication with ambient temperature and expansion volume **24** is connected to an external cooling load (not shown). Refrigerator operation requires net work input.

Stirling cycle engines have not generally been used in practical applications, and Stirling cycle refrigerators have been limited to the specialty field of cryogenics, due to several daunting engineering challenges to their development. These involve such practical considerations as efficiency, vibration, lifetime, and cost. The instant invention addresses these considerations.

A major problem encountered in the design of certain engines, including the compact Stirling engine, is that of the friction generated by a sliding piston resulting from misalignment of the piston in the cylinder and lateral forces exerted on the piston by the linkage of the piston to a rotating crankshaft. In a typical prior art piston-crankshaft configuration such as that depicted in Fig. 2, a piston **10** executes reciprocating motion along longitudinal direction **12** within cylinder **14**. Piston **10** is coupled to an end of connecting rod **16** at a pivot such as a pin **18**. The other end **20** of connecting rod **16** is coupled to a crankshaft **22** at a fixed distance **24** from the axis of rotation **26** of the crankshaft. As crankshaft **22** rotates about the axis of rotation **26**, the connecting rod end **20** connected to the crankshaft traces a circular path while the connecting rod end **28** connected to the piston **10** traces a linear path **30**. The connecting rod angle **32**, defined by the connecting rod longitudinal axis **34** and the axis **30** of the piston, will vary as the crankshaft rotates. The maximum connecting rod angle will depend on the connecting rod offset on the crankshaft

and on the length of the connecting rod. The force transmitted by the connecting rod may be decomposed into a longitudinal component **38** and a lateral component **40**, each acting through pin **18** on piston **10**. Minimizing the maximum connecting rod angle **32** will decrease the lateral forces **40** on the piston and thereby reduce friction and increase the mechanical efficiency of the engine. The maximum connecting rod angle can be minimized by decreasing the connecting rod offset **24** on the crankshaft **22** or by increasing the connecting rod length. However, decreasing the connecting rod offset on the crankshaft will decrease the stroke length of the piston and result in less $\Delta(pV)$ work per piston cycle. Increasing the connecting rod length can not reduce the connecting rod angle to zero but does increase the size of the crankcase resulting in a less portable and compact engine.

Referring now to the prior art engine configuration of Fig. 3, it is known that in order to reduce the lateral forces on the piston, a guide link **42** may be used as a guidance system to take up lateral forces while keeping the motion of piston **10** constrained to linear motion. In a guide link design, the connecting rod **16** is replaced by the combination of guide link **42** and a connecting rod **16**. Guide link **42** is aligned with the wall **44** of piston cylinder **14** and is constrained to follow linear motion by two sets of rollers or guides, forward rollers **46** and rear rollers **48**. The end **50** of guide link **42** is connected to connecting rod **16** which is, in turn, connected to crankshaft **22** at a distance offset from the rotational axis **26** of the crankshaft. Guide link **42** acts as an extension of piston **10** and the lateral forces on the piston that would normally be transmitted to cylinder walls **44** are instead taken up by the two sets of rollers **46** and **48**. Both sets of rollers **46** and **48** are required to maintain the alignment of guide link **42** and to take up the lateral forces being transmitted to the guide link by the connecting rod. The distance **d** between the forward set of rollers and the rear set of rollers

may be reduced to decrease the size of the crankcase (not shown). However, reducing the distance between the rollers will increase the lateral load **54** on the forward set of rollers since the rear roller set acts as a fulcrum **56** to a lever **58** defined by the connection point **52** of the guide link and connecting rod **16**.

The guide link will generally increase the size of the crankcase because the guide link must be of sufficient length that when the piston is at its maximum extension into the piston cylinder, the guide link extends beyond the piston cylinder so that the two sets of rollers maintain contact and alignment with the guide link.

Summary of the Invention

In accordance with one aspect of the invention, in one of its embodiments, there is provided a linkage for coupling a piston undergoing reciprocating linear motion along a longitudinal axis to a crankshaft undergoing rotary motion about a rotation axis of the crankshaft. The longitudinal axis and the rotation axis are substantially orthogonal to each other. The linkage has a guide link with a first end proximal to the piston and coupled to the piston, and a second end distal to the piston such that the rotation axis is disposed between the proximal end and the distal end of the guide link. The linkage has a connecting rod with a connecting end and a crankshaft end, the connecting end rotatably connected to the end of the guide link distal to the piston at a rod connection point and the crankshaft end coupled to the crankshaft at a crankshaft connection point offset from the rotation axis of the crankshaft. Finally, the linkage has a guide link guide assembly for supporting lateral loads at the distal end of the guide link. The guide link guide assembly may include a first roller having a center of rotation fixed with respect to the rotation of the crankshaft and a rim in rolling contact with the distal end of the guide link.

In accordance with alternate embodiments of the present invention, a spring mechanism may be provided for urging the rim of the first roller into contact with the distal end of the guide link. In a further embodiment, the guide link guide assembly may include a second roller in opposition to the first roller, the second roller having a center of rotation and a rim in rolling contact with the distal end of the guide link. The second roller may further include a precision positioner to position of the center of rotation of the second roller with respect to the longitudinal axis. In a preferred embodiment, the precision positioner is a vernier mechanism having an eccentric shaft for varying a distance between the center of rotation of the second roller and the longitudinal axis. The ends of the guide link may be formed of different materials and may be detached for replacement of a worn end.

In accordance with another aspect of the present invention, a machine is provided that has a piston with a longitudinal travel axis and a crankshaft capable of rotation about a rotation axis, the rotation axis being substantially orthogonal to the longitudinal axis. The machine has a guide link having a length and a first end proximal to the piston and coupled to the piston and a second end that is distal to the piston such that the rotation axis is disposed between the proximal end and the distal end of the guide link. The machine has a connecting rod with a connecting end and a crankshaft end, the connecting end rotatably connected to the end of the guide link distal to the piston and the crankshaft end coupled to the crankshaft at a crankshaft connection point offset from the rotation axis of the crankshaft. Finally, the guide link is constrained to follow a substantially linear path at a discrete number of points along its length.

In accordance with yet another aspect of the present invention, an improvement is provided to a Stirling cycle machine of the type wherein a displacer piston undergoes reciprocating motion along a first longitudinal axis and a compression piston undergoes

reciprocating motion along a second longitudinal axis. As used in this description and the following claims, a displacer piston is either a piston without a seal or a piston with a seal (commonly known as an "expansion" piston). The improvement has a folded guide link linkage for coupling each of the pistons to a crankshaft. In a further embodiment, the improvement has a guide link guide assembly with precision positioning. In another further embodiment, an improvement consists of a crankshaft coupling assembly for coupling a first connection rod and a second connection rod to the crankshaft such that the reciprocating motion along the first and second longitudinal axes are substantially coplanar. The crankshaft coupling assembly may be a "fork and blade" type assembly.

In accordance with another aspect of the invention, another improvement is provided to a Stirling cycle engine. The improvement has a bearing mount coupled to at least one support bracket which is coupled to a pressure enclosure such that a dimensional change of the pressure enclosure is substantially decoupled from the bearing mount. In another embodiment, a method for aligning a piston in a cylinder, the piston undergoing reciprocating motion along a longitudinal axis and coupled to a guide link having a length, comprises providing a first guide element along the length of the guide link, the first guide element having a spring mechanism for urging the guide element into contact with the guide link and providing a second guide element along the length of the guide link, the second guide element in opposition to the first guide element and having a precision positioner for positioning the second guide element with respect to the longitudinal axis. In a preferred embodiment, the precision positioner is a vernier mechanism having an eccentric shaft for varying a distance between the second guide element and the longitudinal axis.

In another further embodiment, an alignment device is provided having a first guide element located along the length of the guide link, the first guide element having a spring

mechanism for urging the guide element into contact with the guide link and a second guide element in opposition to the first guide element, the second guide element having a precision positioner for positioning the second guide element with respect to the longitudinal axis.

Brief Description of the Drawings

The invention will be more readily understood by reference to the following description, taken with the accompanying drawings, in which:

FIGS **1a-1e** depict the principle of operation of a prior art Stirling cycle machine.

FIG. **2** is a cross-sectional view of a prior art linkage for an engine;

FIG. **3** is a cross-sectional view of a second prior art linkage for an engine, the linkage having a guide link;

FIG. **4** is a cross-sectional view of a folded guide link linkage for an engine in accordance with a preferred embodiment of the present invention;

FIG. **5a** is a cross-sectional view of a piston and guide assembly for allowing the precision alignment of piston motion using vernier alignment in accordance with a preferred embodiment of the invention.

FIG. **5b** is a side view of the precision alignment mechanism in accordance with an embodiment of the invention.

FIG. **5c** is a perspective view of the precision alignment mechanism of Figure 5b in accordance with an embodiment of the invention.

FIG. **5d** is a top view of the precision alignment mechanism of Figure 5b in accordance with an embodiment of the invention.

FIG. **5e** is a top view of the precision alignment mechanism of Figure 5b with both the locking holes and the bracket holes showing in accordance with an embodiment of the

invention.

FIG. **6** is a cross-sectional view of a folded guide link linkage for a two-piston machine such as a Stirling cycle machine in accordance with a preferred embodiment of the present invention;

FIG. **7** is a cross-sectional view of a "fork-and blade" type crankshaft coupling assembly in accordance with a preferred embodiment of the invention.

FIG. **8** is a perspective view of one embodiment of the dual folded guide link linkage of Fig. 6.

FIG. **9a** is a perspective view of a Stirling engine in accordance with a preferred embodiment of the invention.

FIG. **9b** is a perspective view of the cold section base plate and the lower bracket of Figure 9a where the lower bracket is mounted on the cold section base plate in accordance with a preferred embodiment of the invention.

Detailed Description of Preferred Embodiments

Referring now to FIG. 4, a schematic diagram is shown of a folded guide link linkage designated generally by numeral **100**. A piston **101** is rigidly coupled to the piston end of a guide link **103** at a piston connection point **102**. Guide link **103** is rotatably connected to a connecting rod **105** at a rod connection point **104**. The piston connection point **102** and the rod connection point **104** define the longitudinal axis **120** of guide link **103**.

Connecting rod **105** is rotatably connected to a crankshaft **106** at a crankshaft connection point **108** which is offset a fixed distance from the crankshaft axis of rotation **107**. The crankshaft axis of rotation **107** is orthogonal to the longitudinal axis **120** of the guide link **103** and the crankshaft axis of rotation **107** is disposed between the rod connection point **104**

and the piston connection point **102**. In a preferred embodiment, the crankshaft axis of rotation **107** intersects the longitudinal axis **120**.

An end **114** of guide link **103** is constrained between a first roller **109** and an opposing second roller **111**. The centers of roller **109** and roller **111** are designated respectively by numerals **110** and **112**. The position of guide link piston linkage **100** depicted in Fig. 4 is that of mid-stroke point in the cycle. This occurs when the radius **116** between the crankshaft connection point **108** and the crankshaft axis of rotation **107** is orthogonal to the plane defined by the crankshaft axis of rotation **107** and the longitudinal axis of the guide link **103**. In a preferred embodiment, the rollers **109**, **111** are placed with respect to the guide link **103** in such a manner that the rod connection point **104** is in the line defined by the centers **110**, **112** of the rollers **109**, **111** at mid-stroke. As rollers **109**, **111** wear during use, the misalignment of the guide link will increase. In a preferred embodiment, the first roller **109** is spring loaded to maintain rolling contact with the guide link **103**. In accordance with embodiments of the invention, guide link **103** may comprise subcomponents such that the portion **113** of the guide link proximal to the piston may be a lightweight material such as aluminum, whereas the "tail" portion **114** of the guide link distal to the piston may be a durable material such as steel to reduce wear due to friction at rollers **109** and **111**.

Alignment of the longitudinal axis **120** of the guide link **103** with respect to piston cylinder **14** is maintained by the rollers **109**, **111** and by the piston **101**. As crankshaft **106** rotates about the crankshaft axis of rotation **107**, the rod connection point **104** traces a linear path along the longitudinal axis **120** of the guide link **103**. Piston **101** and guide link **103** form a lever with the piston **101** at one end of the lever and the rod end **114** of the guide link **103** at the other end of the lever. The fulcrum of the lever is on the line defined by the

centers **110, 112** of the rollers **109, 111**. The lever is loaded by a force applied at the rod connection point **104**. As rod connection point **104** traces a path along the longitudinal axis of the guide link **103**, the distance between the rod connection point **104** and the fulcrum, the first lever arm, will vary from zero to one-half the stroke distance of the piston **101**. The second lever arm is the distance from the fulcrum to the piston **101**. The lever ratio of the second lever arm to the first lever arm will always be greater than one, preferably in the range from 5 to 15. The lateral force at the piston **101** will be the forced applied at the rod connection point **104** scaled by the lever ratio; the larger the lever ratio, the smaller the lateral force at the piston **101**.

By moving the connection point to the side of the crankshaft axis distal to that of the piston, the distance between the crankshaft axis and the piston cylinder does not have to be increased to accommodate the roller housing. Additionally, only one set of rollers is required for aligning the piston, thereby advantageously reducing the size of the roller housing and the overall size of the engine. In accordance with the invention, while the piston experiences a non-zero lateral force (unlike a standard guide link design where the lateral force of a perfectly aligned piston is zero), the lateral force can be at least an order of magnitude less than that experienced by a simple connecting rod crankshaft arrangement due to the large lever arm created by the guide link.

Lateral forces on a piston can give rise to noise and to wear. Additional friction may be generated by the misalignment of the piston in the cylinder. A solution to the alignment problem is now discussed with reference to Figures 5a-5e. Figure 5a shows a schematic diagram of a piston **201** and a guide assembly **209** for allowing precision alignment of piston motion using vernier alignment in accordance with a preferred embodiment of the invention. The piston **201** executes a reciprocating motion along a longitudinal axis **202** in cylinder **200**.

A guide link **204** is coupled to the piston **201**. An end of the guide link **204** is constrained between a first roller **205** and an opposing second roller **207**. The centers of roller **205** and roller **207** are designated respectively by numerals **206** and **208**. A piston guide ring **203** may be used at one end of the piston **201** to prevent piston **201** from touching the cylinder **200**. However, if piston **201** is not aligned to move in a straight line along longitudinal axis **202**, it is possible other points along the length of piston **201** not coupled to the guide ring may contact the cylinder **200**. In a preferred embodiment, piston **201** is aligned using rollers **205** and **207** and guide link **204** such that piston **201** moves along the longitudinal axis **202** in a straight line and is substantially centered with respect to cylinder **200**.

In accordance with a preferred embodiment of the invention, the piston **201** may be aligned with respect to the piston cylinder **200** by adjusting the position of the center **208** of the second roller **207**. The first roller **205** is spring loaded to maintain rolling contact with the guide link **204**. The second roller **207** is mounted on an eccentric flange such that rotation of the flange causes the second roller **207** to move laterally with respect to longitudinal axis **202**. A single pin (not shown) may be used to secure the second roller **207** into a position. The movement of the second roller **207** will cause the guide link **204** and the piston **201** to also move laterally with respect to the longitudinal axis **202**. In this manner, the piston **201** may be aligned so as to move in cylinder **200** in a straight line which is substantially centered with respect to cylinder **200**.

Figure 5b shows a side view of one embodiment of a precision alignment mechanism. A roller **207** is rotatably mounted on a locking eccentric **211** having a lower end **212** and an upper end **213**. The roller is mounted on a portion **210** of the locking eccentric **211** having a roller axis of rotation that is offset from the axis of rotation of the locking eccentric **211**. The

lower end **212** is rotatably mounted in a lower bracket (not shown). The upper end **213** is rotatably mounted on an upper bracket **214**. Figure 5c shows a perspective view of the embodiment shown in Figure 5b. The upper bracket **214** has a plurality of bracket holes **220** drilled through the upper bracket **214**. In a preferred embodiment, eighteen bracket holes are drilled through the upper bracket **214**. The bracket holes **220** are offset a distance from the axis of rotation of the locking eccentric **211** and are evenly spaced around the circumference defined by the offset distance.

Figure 5d shows a the top view of the embodiment shown in Figure 5b. The upper end **213** of the locking eccentric **211** has a plurality of locking holes **215**. The number of locking holes **215** should not be identical to the number of bracket holes **220**. In a preferred embodiment, the number of locking holes **215** is nineteen. The locking holes **215** are offset from the axis of rotation of the locking eccentric **211** by the same distance used to offset the bracket holes **220**. The locking holes **215** are evenly spaced around the circumference defined by the offset distance. Figure 5d also shows a locking nut **216** that allows the locking eccentric **211** to rotate when the locking nut **216** is loose. When the locking nut **216** is tightened, the locking nut **216** makes a rigid connection between the locking eccentric **211** and the upper bracket **214**. Figure 5e is the same view as shown in Figure 5d but with the locking holes **215** shown.

During assembly, the piston is aligned in the following manner. The folded guide link is assembled with the locking nut **216** in a loosened state. The piston **201** (Figure 5a) is aligned within the piston cylinder **200** (Figure 5a) visually by rotating the locking eccentric **211**. As the locking eccentric **211** is rotated, the roller axis of rotation **208** (Figure 5a) will be displaced both laterally and longitudinally to the guide link longitudinal axis **202** (Figure 5a).

The large lever ratio of the present invention requires only a very small displacement of the roller axis of rotation **208** (Figure 5a) with respect to the longitudinal axis **202** (Figure 5a) to align the piston **201** (Figure 5a) within the piston cylinder **200** (Figure 5a). In accordance with an embodiment of the invention, the maximum displacement range may be from 0.000 inches to 0.050 inches. In a preferred embodiment, the maximum displacement is between 0.010 inches and 0.030 inches. As the locking eccentric **211** is rotated, the locking holes **215** will align with a bracket hole **220**. Figure 5d indicates such an alignment **230**. Once the piston **201** (Figure 5a) is aligned in the piston cylinder **200** (Figure 5a), a pin (not shown) is inserted through the aligned bracket hole and into the aligned locking hole thereby locking the locking eccentric **211**. The locking nut **216** is then tightened to rigidly connect the upper bracket **214** to the locking eccentric **211**.

In accordance with a preferred embodiment of the invention, a dual folded guide link piston linkage such as shown in cross-section in Fig. 6 and designated there generally by numeral **300** may be incorporated into a compact Stirling engine. Referring now to FIG. 6, pistons **301** and **311** are the displacer and compression pistons, respectively, of a Stirling cycle engine. As used in this description and the following claims, a displacer piston is either a piston without a seal or a piston with a seal (commonly known as an "expansion" piston). The Stirling cycle is based on two pistons executing reciprocating linear motion about 90° out of phase with one another. This phasing is achieved when the pistons are oriented at right angles and the respective connecting rods share a common pin of a crankshaft. Additional advantages of this orientation include reduction of vibration and noise. Additionally, the two pistons may advantageously lie in the same plane to eliminate shaking vibrations orthogonal to the plane of the pistons. In accordance with a preferred embodiment, a "fork and blade" type crankshaft coupling assembly, as described below, is used to couple the connecting rods

306 and 316 to the crankshaft 308 at crankshaft connection points 307 and 317 respectively so that the pistons 301 and 311 may move in the same plane.

Figure 7 is a cross-sectional view of a "fork and blade" type coupling assembly. A crankshaft 400 has a crankshaft pin 401. The crankshaft pin 401 rotates about the crankshaft axis of rotation 402. A first coupling element 403 is a "blade" link. In other words, as seen in Figure 7, the "blade" is a single link used to couple a first connecting rod to the crankshaft pin 401. A second coupling element 404 includes a "fork" link. The "fork", as seen in Figure 7, is a pair of links used to couple a second connecting rod to the crankshaft pin 401. The first and second coupling elements 403 and 404 may be used to couple two connecting rods to the same crankshaft pin such that the motion of the connecting rods is substantially within the same plane. Referring again to Figure 6, a "fork and blade" type crankshaft coupling assembly, as shown in Figure 7, may be used to connect the first coupling rod 306 and the second coupling rod 316 to the crankshaft 308 at crankshaft connection points 307 and 317 respectively. While the invention is described generally with reference to the Stirling engine shown in FIG. 6, it is to be understood that many engines as well as refrigerators may similarly benefit from various embodiments and improvements which are subjects of the present invention.

The configuration of a Stirling engine shown in FIG. 6 in cross-section, and in perspective in FIG. 8, is referred to as an alpha configuration, characterized in that compression piston 311 and displacer piston 301 undergo linear motion within respective and distinct cylinders: compression piston 311 in compression cylinder 320 and displacer piston 301 in expansion cylinder 322. Guide link 303 and guide link 313 are rigidly coupled to displacer piston 301 and compression piston 311 at piston connection points 302 and 312

respectively. Connecting rods **306** and **316** are rotationally coupled at connection points **305** and **315** of the distal ends of guide links **303** and **313** to crankshaft **308** at crankshaft connection points **307** and **317**. Lateral loads on guide links **303** and **313** are taken up by roller pairs **304** and **314**. As discussed above with respect to Figures 4 and 5, the pistons **301** and **311** may be aligned within the cylinders **320** and **322** respectively such using precision alignment of roller pairs **304** and **314**.

As described above with respect to Figures 1a-1f, a Stirling engine operates under pressurized conditions. Typically, a crankcase is used to support the crankshaft and maintain the pressurized conditions under which the Stirling engine operates. The crankshaft would be supported at both ends by crankshaft bearing mounts which would be mounted in the crankcase itself. As the crankcase is pressurized, however, the dimensions of the crankcase may change or deform. If the same structure is used to support the crankshaft, the deformation of the crankcase may result in a misalignment of the crankshaft which places a tremendous burden on the bearings and significantly reduces the lifetime of the engine. In order to reduce or prevent the misalignment of the crankshaft caused by the deformation of the crankcase, the support function of the crankcase may be separated from the pressure function of the crankcase as shown in Figure 9a.

Figure 9a is a perspective view of a Stirling engine in accordance with a preferred embodiment of the invention. A piston guide link **503** and roller **507** assembly is shown as described with respect to Figures 4, 7 and 8. A cold section base plate **501** is coupled to a pressure enclosure **504** to form a crankcase and to define a pressurized volume. An upper bracket **506** and a lower bracket **505** are attached to the cold section base plate **501** using bracket mounting holes **509** on the bracket base mount **502** of the cold section base plate **501**. In a preferred embodiment, the upper bracket **506** and the lower bracket **505** are attached to

the cold section base plate **501** using screws. A crankshaft **508** is supported on both ends by crankshaft bearing mounts (not shown). The crankshaft bearing mounts are mounted on the upper bracket **506** and the lower bracket **505**. In this manner, the bearing mounts are advantageously not directly mounted on the crankcase. The roller **507** is also coupled to the upper bracket **506** and the lower bracket **505** as described with respect to Figures 5a-5e.

Figure 9b is a perspective view of the cold section base plate **501** coupled to the lower bracket **505** of Figure 9a. The crankshaft **508** is connected to the lower bracket **505**. The lower bracket **505** is mounted on the cold section base plate **501**. An opening **510** in the cold section base plate **501** is provided for a piston and a cylinder. As described above, in a preferred embodiment, the crankshaft **508** is supported by crankshaft bearing mounts (not shown) at both ends. The bearing mounts are then mounted on the upper **506** and lower **505** brackets. This configuration advantageously decouples the deformation of the crankcase caused by the pressurized operating conditions of the Stirling engine from the engine alignment. While the crankcase will still deform under high pressure, the deformation will not affect the alignment of the crankshaft because the crankshaft is not directly mounted on the crankcase. This configuration also advantageously reduces the bearing loads by shortening the distance between the bearing mounts (the distance between the upper and lower brackets instead of the distance between the opposite faces of the crankcase). In a preferred embodiment, the region of the cold base plate may also be locally reinforced to further reduce the local deformation of the bracket mount due to the pressurized operating conditions.

The devices and methods described herein may be applied in other applications besides the Stirling engine in terms of which the invention has been described. The described embodiments of the invention are intended to be merely exemplary and numerous variations

and modifications will be apparent to those skilled in the art. All such variations and modifications are intended to be within the scope of the present invention as defined in the appended claims.

WE CLAIM:

1. A linkage for coupling a piston undergoing reciprocating linear motion along a longitudinal axis to a crankshaft undergoing rotary motion about a rotation axis of the crankshaft, the longitudinal axis and the rotation axis being substantially orthogonal to each other, the linkage comprising:

a guide link having a first end proximal to the piston, the first end coupled to the piston, and having a second end distal to the piston such that the rotation axis is disposed between the proximal end and the distal end of the guide link;

a connecting rod having a connecting end and a crankshaft end, the connecting end rotatably connected to the end of the guide link distal to the piston at a rod connection point and the crankshaft end coupled to the crankshaft at a crankshaft connection point offset from the rotation axis of the crankshaft; and

a guide link guide assembly for supporting lateral loads at the distal end of the guide link, the guide link guide assembly having a first roller, the first roller having a center of rotation fixed with respect to the rotation axis of the crankshaft and a rim in rolling contact with the distal end of the guide link.

2. A linkage according to claim 1, wherein the guide link guide assembly further includes a spring mechanism for urging the rim of the first roller into contact with the distal end of the guide link.

3. A linkage according to claim 2, wherein the guide link guide assembly further includes a second roller in opposition to the first roller, the second roller having a center of rotation and a rim in rolling contact with the distal end of the guide link.

4. A linkage according to claim 3, wherein the second roller further includes a precision positioner to position the center of rotation of the second roller with respect to the longitudinal axis.

5. A linkage according to claim 4, wherein the precision positioner is a vernier mechanism having an eccentric shaft for varying the distance between the center of rotation of the second roller and the longitudinal axis.

6. A linkage according to claim 1, wherein a line defined by the centers of the first and second rollers includes the rod connection point when the crankshaft connection point is at a mid-stroke position.

7. A machine comprising:

a piston having a longitudinal travel axis;

a crankshaft capable of rotation about a rotation axis, the rotation axis being substantially orthogonal to the longitudinal axis;

a guide link having a length and a first end proximal to the piston, the first end coupled to the piston, the guide link having a second end distal to the piston such that the rotation axis is disposed between the proximal end and the distal end of the guide link; and

a connecting rod having a connecting end and a crankshaft end, the connecting end rotatably connected to the end of the guide link distal to the piston and the crankshaft end coupled to the crankshaft at a crankshaft connection point offset from the rotation axis of the crankshaft;

wherein the guide link is constrained to follow a substantially linear path at a discrete

number of points along its length.

8. A guide link for coupling a piston undergoing reciprocating linear motion along a longitudinal axis to a crankshaft undergoing rotary motion about a rotation axis of the crankshaft, the longitudinal axis and the rotation axis being substantially orthogonal to each other, the guide link comprising:

a first end proximal to the piston, the first end coupled to the piston; and

a second end distal to the piston and coupled to the crankshaft at a point displaced from the rotation axis such that the rotation axis is disposed between the first end and the second end of the guide link.

9. A guide link according to claim 8, further including a coupling for connecting the first end to the second end such that the first end may be decoupled from the second end for replacement of a worn second end.

10. In a Stirling cycle machine of the type wherein a displacer piston undergoes reciprocating motion along a first longitudinal axis and a compression piston undergoes reciprocating motion along a second longitudinal axis, the improvement comprising:

a crankshaft undergoing rotary motion about a rotation axis of the crankshaft for coupling mechanical energy with respect to the machine;

a first and a second guide link, the first guide link having a first end proximal to the displacer piston and coupled to the displacer piston, the second guide link having a first end proximal to the compression piston and coupled to the compression piston, each guide link having a second end distal to the respective piston such each rotation axis is disposed

between the proximal end of the respective guide link and the distal end of the respective guide link;

two connecting rods, each connecting rod having a connecting end and a crankshaft end, the connecting end rotatably connected to the end of one of the guide links distal to the respective piston at a rod connection point and the crankshaft end coupled to the crankshaft at a crankshaft connection point offset from the rotation axis of the crankshaft; and

two guide link guide assemblies, each guide link guide assembly in contact with the distal end of one of the guide links and for supporting lateral loads at the distal ends of the guide links.

11. In the Stirling cycle machine of claim 10, the improvement wherein each guide link guide assembly further includes a first roller, the first roller having a center of rotation fixed with respect to the rotation axis of the crankshaft and having a rim in contact with the distal end of the respective guide link.

12. In the Stirling cycle machine of claim 11, the improvement wherein each guide link guide assembly further includes a spring mechanism for urging the rim of the first roller into contact with the distal end of the respective guide link.

13. In the Stirling cycle engine of claim 12, the improvement wherein each guide link guide assembly further includes a second roller in opposition to the first roller, the second roller having a center of rotation and a rim in rolling contact with the distal end of the guide link.

14. In the Stirling cycle engine of claim 13, the improvement wherein at least one of the second rollers includes a precision positioner to position the center of rotation of the at least one second roller with respect to the respective longitudinal axis.

15. In the Stirling cycle machine of claim 14, the improvement wherein the precision positioner is a vernier mechanism having an eccentric shaft for varying a distance between the center of rotation of the second roller and the respective longitudinal axis.

16. In the Stirling cycle machine of claim 10, the improvement wherein the first and second longitudinal axes are substantially coplanar.

17. In a Stirling cycle machine of the type wherein a displacer piston undergoes reciprocating motion along a first longitudinal axis and a compression piston undergoes reciprocating motion along a second longitudinal axis, the improvement comprising:

a crankshaft undergoing rotary motion about a rotation axis of the crankshaft for coupling mechanical energy with respect to the machine;

a first and second guide link, the first guide link having a first end proximal to the displacer piston and coupled to the displacer piston, the second guide link having a first end proximal to the compression piston and coupled to the compression piston, each guide link having a second end distal to the respective piston such that each rotation axis is disposed between the proximal end of the respective guide link and the distal end of the guide link;

a first connecting rod, the first connecting rod having a connecting end and a crankshaft end, the connecting end rotatably connected to the end of the first guide link distal

to the displacer piston at a rod connection point and the crankshaft end coupled to the crankshaft at a first crankshaft connection point offset from the rotation axis of the crankshaft;

a second connecting rod, the second connecting rod having a connecting end and a crankshaft end, the connecting end rotatably connected to the end of the second guide link distal to the compression piston at a rod connection point and the crankshaft end coupled the crankshaft at a second crankshaft connection point offset from the rotation axis of the crankshaft;

a crankshaft coupling assembly for coupling the first connection rod and the second connection rod to the crankshaft such that the reciprocating motion along the first and second longitudinal axes is substantially coplanar; and

two guide link guide assemblies, each guide link guide assembly in contact with the distal end of one of the guide links and for supporting lateral loads at the distal ends of the guide links.

18. In the Stirling cycle machine of claim 17, the improvement wherein the crankshaft coupling assembly further includes a fork coupling element connected between the first connecting rod and the crankshaft and a blade coupling element connected between the second connecting rod and the crankshaft.

19. In a Stirling cycle machine of the type wherein an displacer piston undergoes reciprocating motion along a first longitudinal axis in a first cylinder and a compression piston undergoes reciprocating motion along a second longitudinal axis in a second cylinder, the pistons being coupled to a crankshaft, the improvement comprising:

a pressure enclosure for containing a working fluid, the working fluid undergoing successive closed cycles of heating, expansion, cooling and compression;

at least one support bracket coupled to the pressure enclosure; and

a bearing mount for supporting the crankshaft, the bearing mount coupled to the support bracket such that a dimensional change of the pressure enclosure is substantially decoupled from the bearing mount.

20. A method for aligning a piston in a cylinder, the piston undergoing reciprocating motion along a longitudinal axis and coupled to a guide link having a length, the method comprising:

providing a first guide element located along the length of the guide link, the first guide element having a spring mechanism for urging the guide element into contact with the guide link;

providing a second guide element in opposition to the first guide element, the second guide element having a precision positioner for positioning the second guide element with respect to the longitudinal axis;

moving the position of the second guide element so as to change the position of the guide link and the piston with respect to the longitudinal axis.

21. A method according to claim 20, wherein the first guide element is a roller having a center of rotation and a rim in rolling contact with the guide link and a second guide element is a roller having a center of rotation and a rim in rolling contact with the guide link.

22. A method according to claim 20, wherein the precision positioner is a vernier.

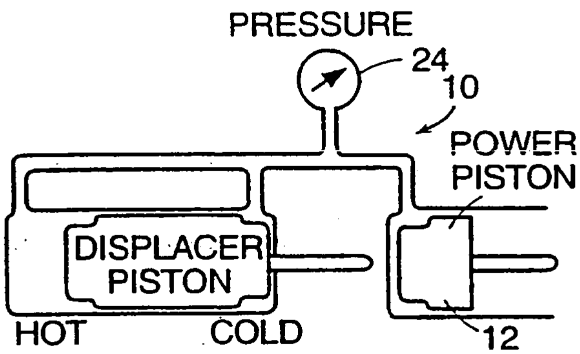
mechanism having an eccentric shaft for varying a distance between the second guide element and the longitudinal axis.

23. An alignment device for aligning a piston in a cylinder, the piston undergoing reciprocating motion along a longitudinal axis coupled to a guide link having a length, the alignment device comprising:

a first guide element located along the length of the guide link, the first guide element having a spring mechanism for urging the guide element into contact with the guide link; and

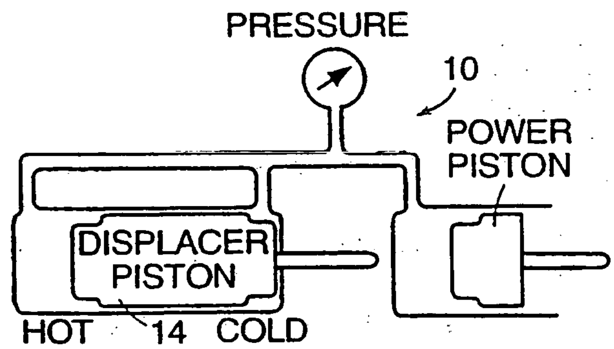
a second guide element in opposition to the first guide element, the second guide element having a precision positioner for positioning the second guide element with respect to the longitudinal axis.

24. The alignment device of claim 23, wherein the precision positioner is a vernier mechanism having an eccentric shaft for varying a distance between the second guide element and the longitudinal axis.



READY FOR START OF EXPANSION STROKE

FIG. 1c
PRIOR ART



END OF EXPANSION STROKE

FIG. 1d
PRIOR ART

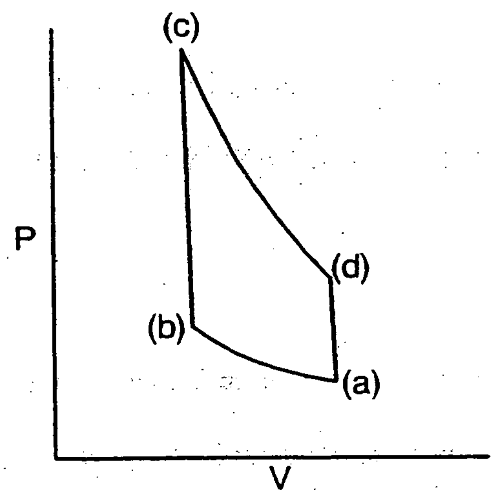
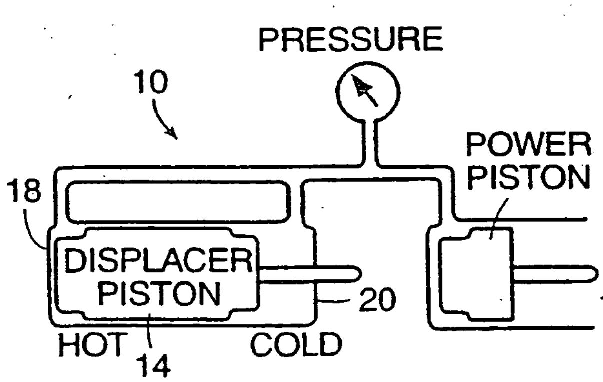
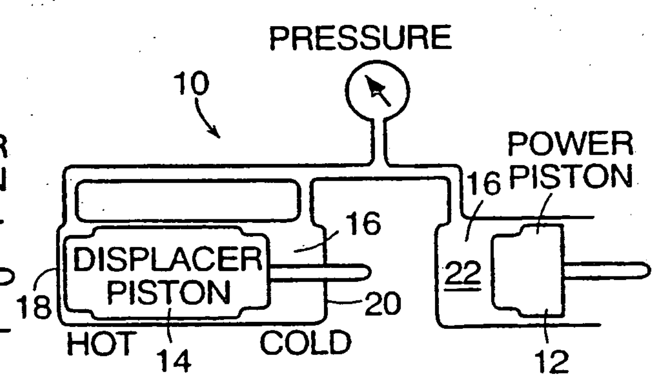


FIG. 1e
PRIOR ART



END OF COMPRESSION STROKE

FIG. 1b
PRIOR ART



READY FOR START OF COMPRESSION STROKE

FIG. 1a
PRIOR ART

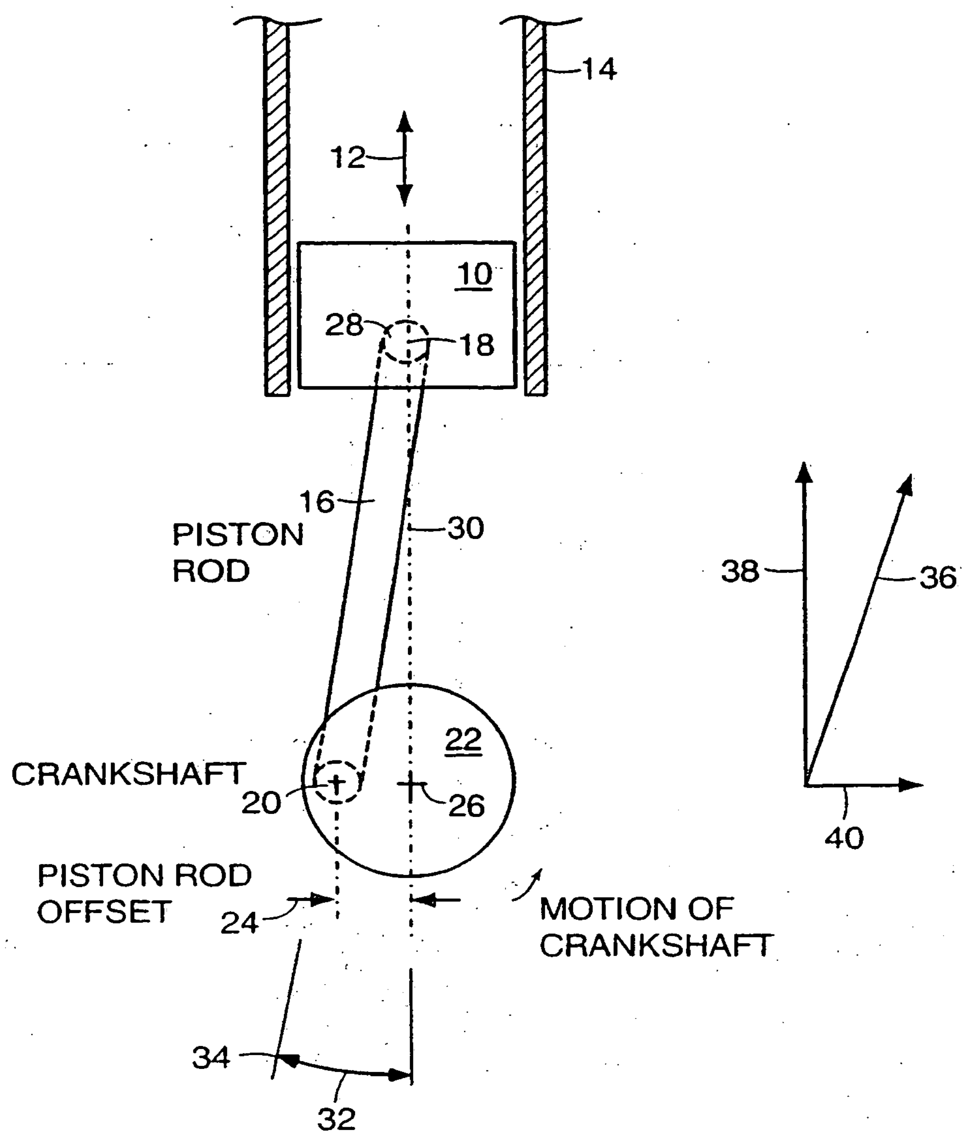


FIG. 2
PRIOR ART

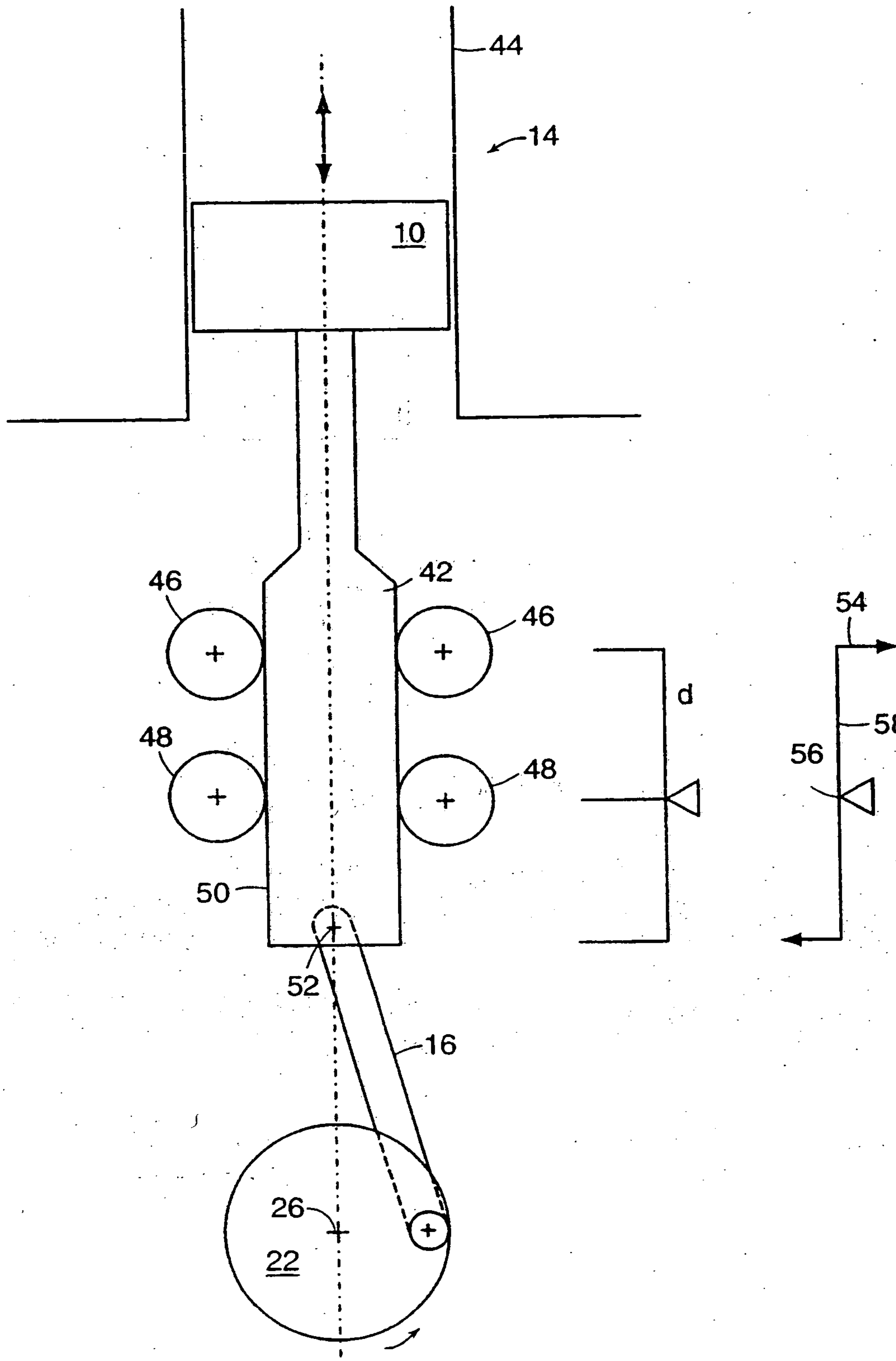


FIG. 3
PRIOR ART

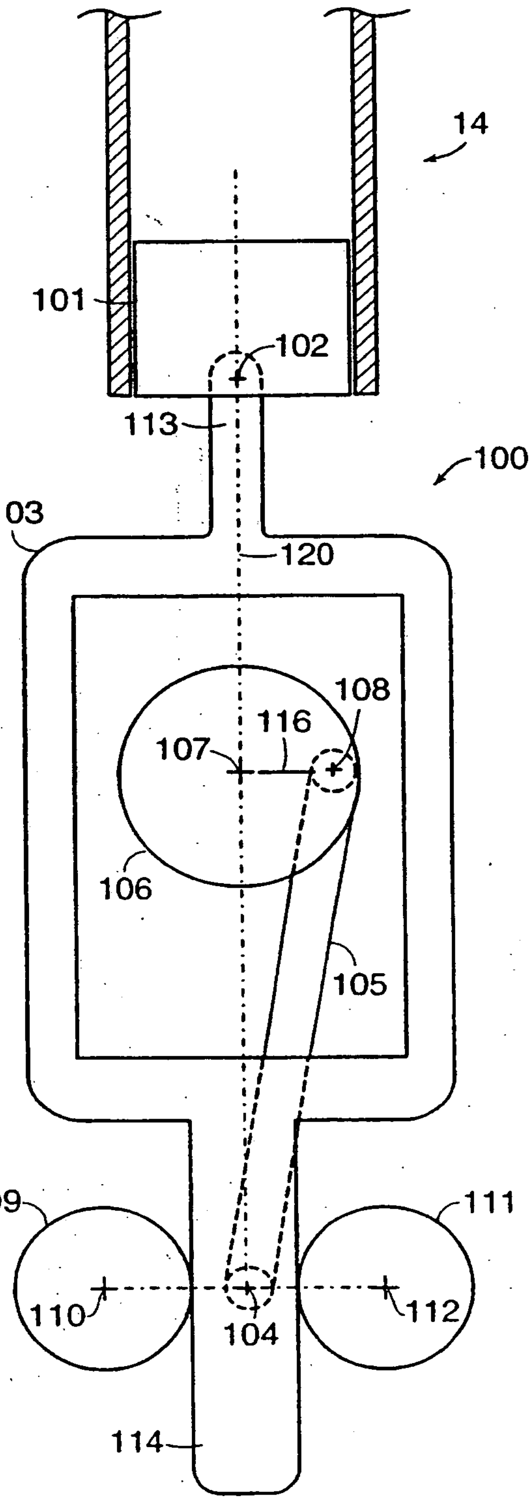

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FIG. 4

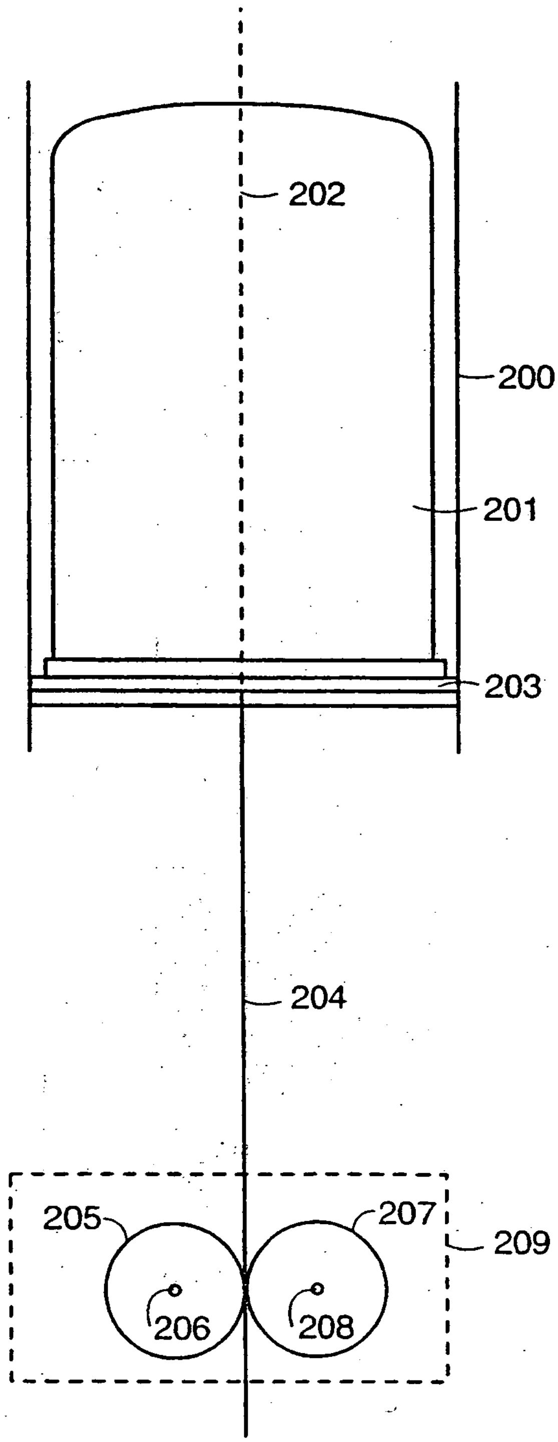

FIG. 5a

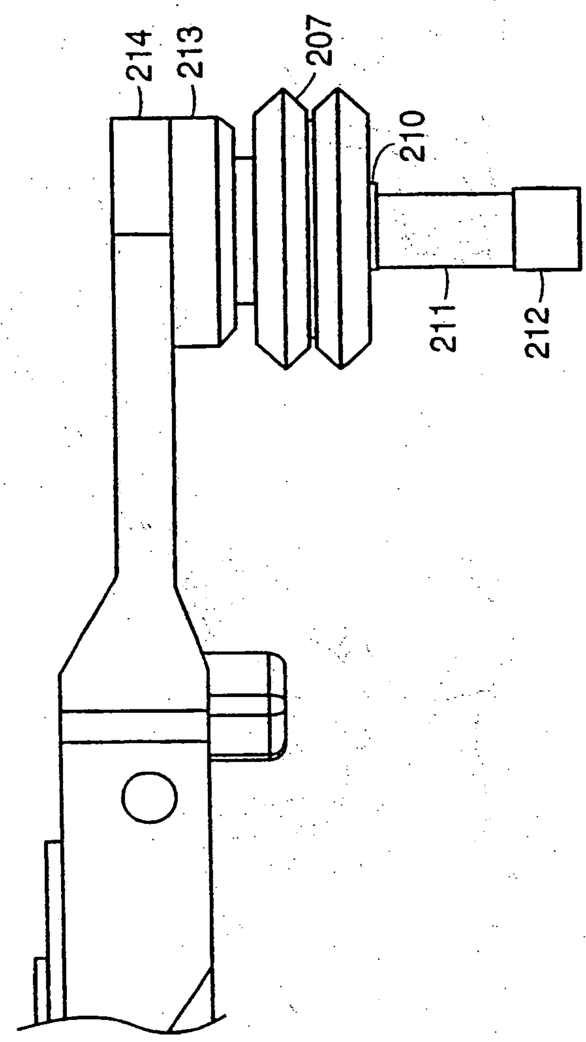

FIG. 5b

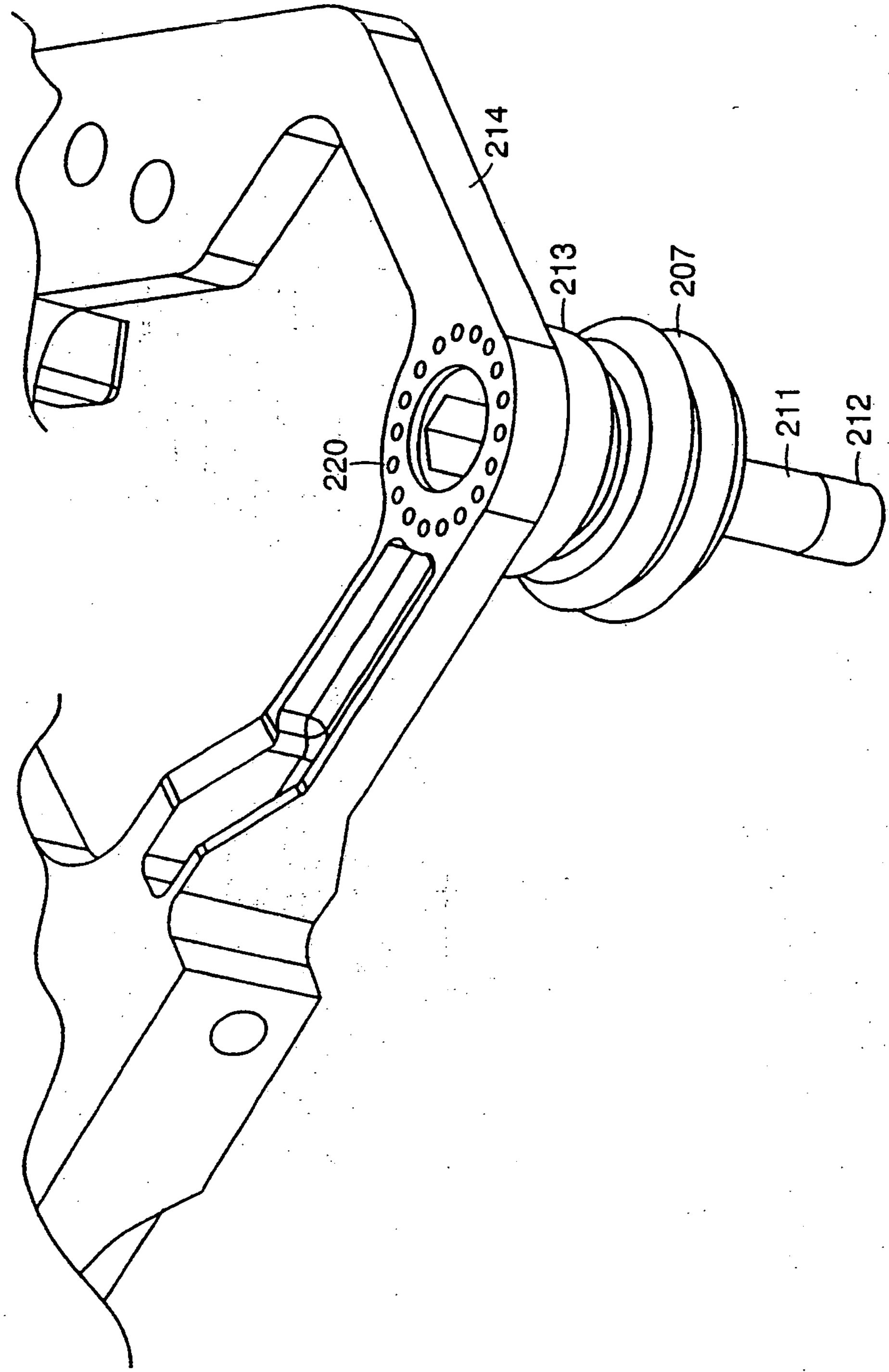

FIG. 5c

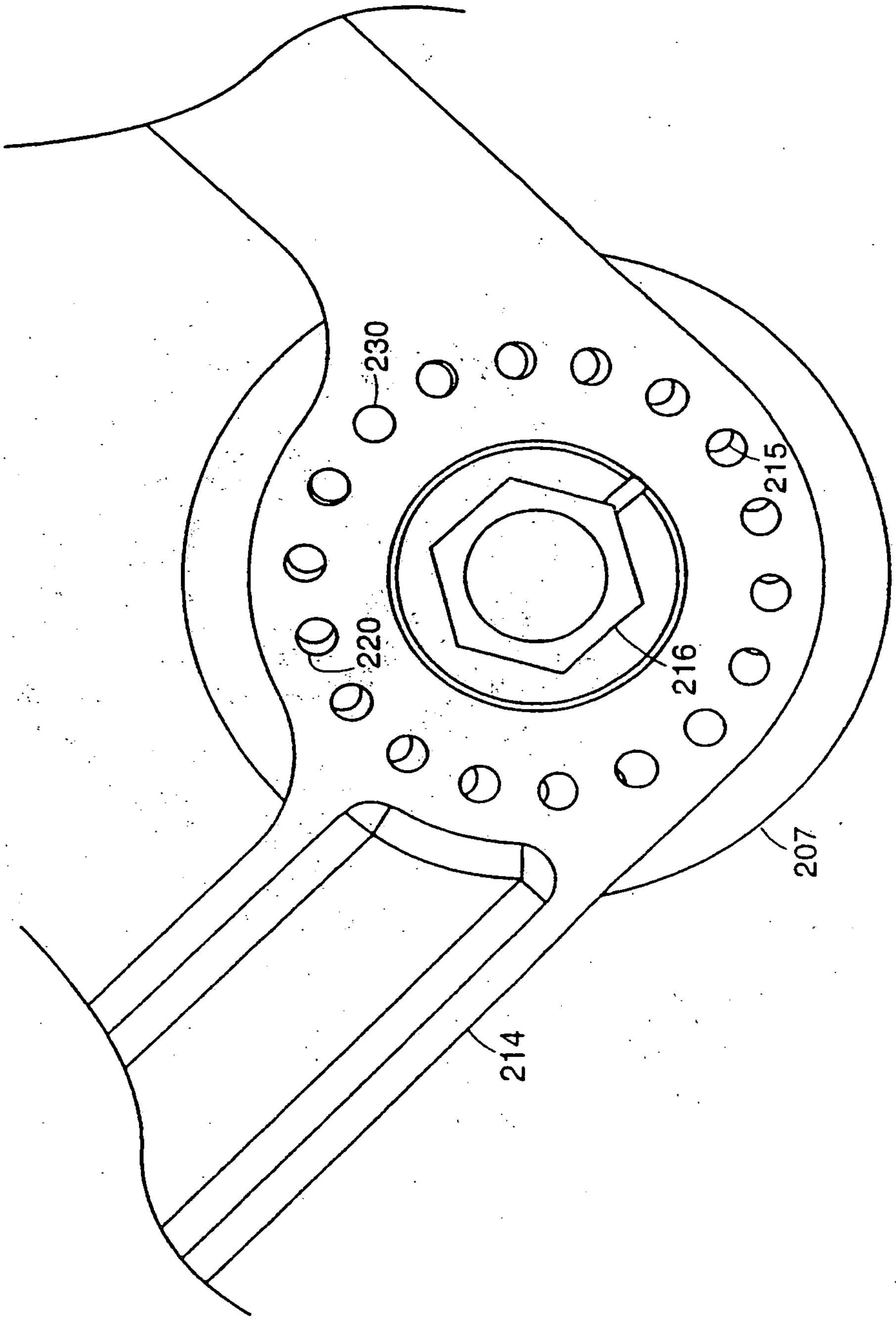

FIG. 5d

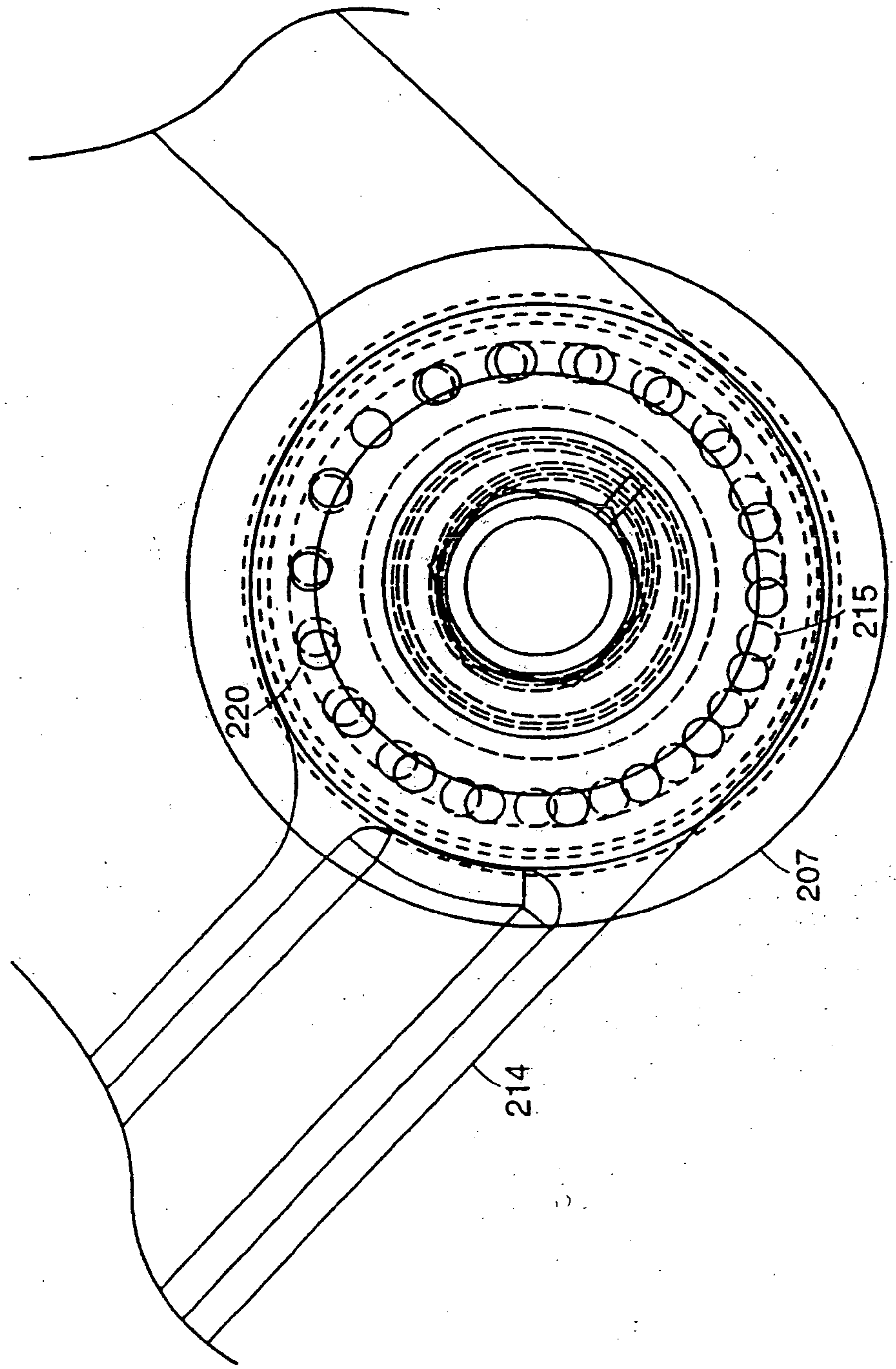

FIG. 5e

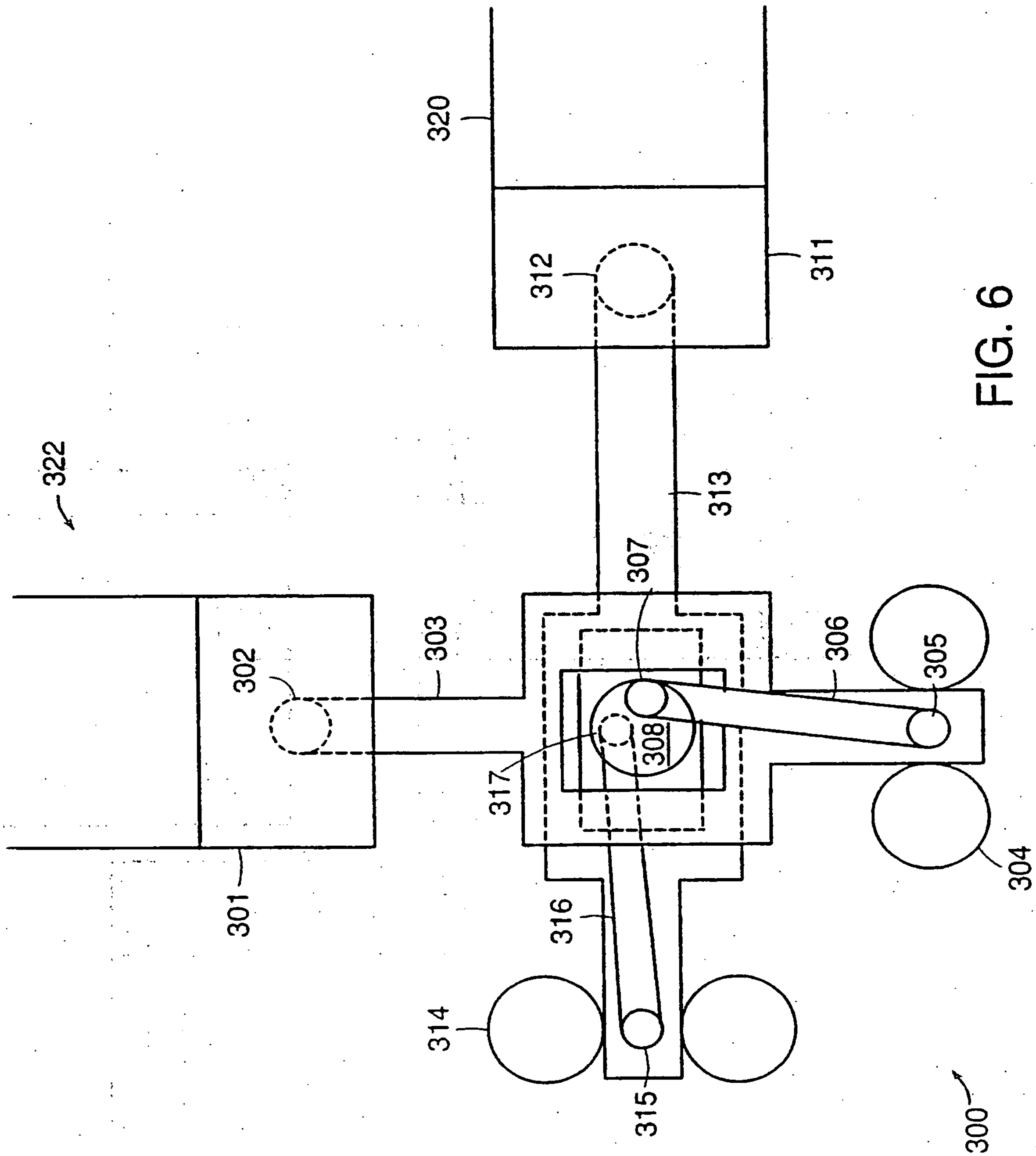
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FIG. 6

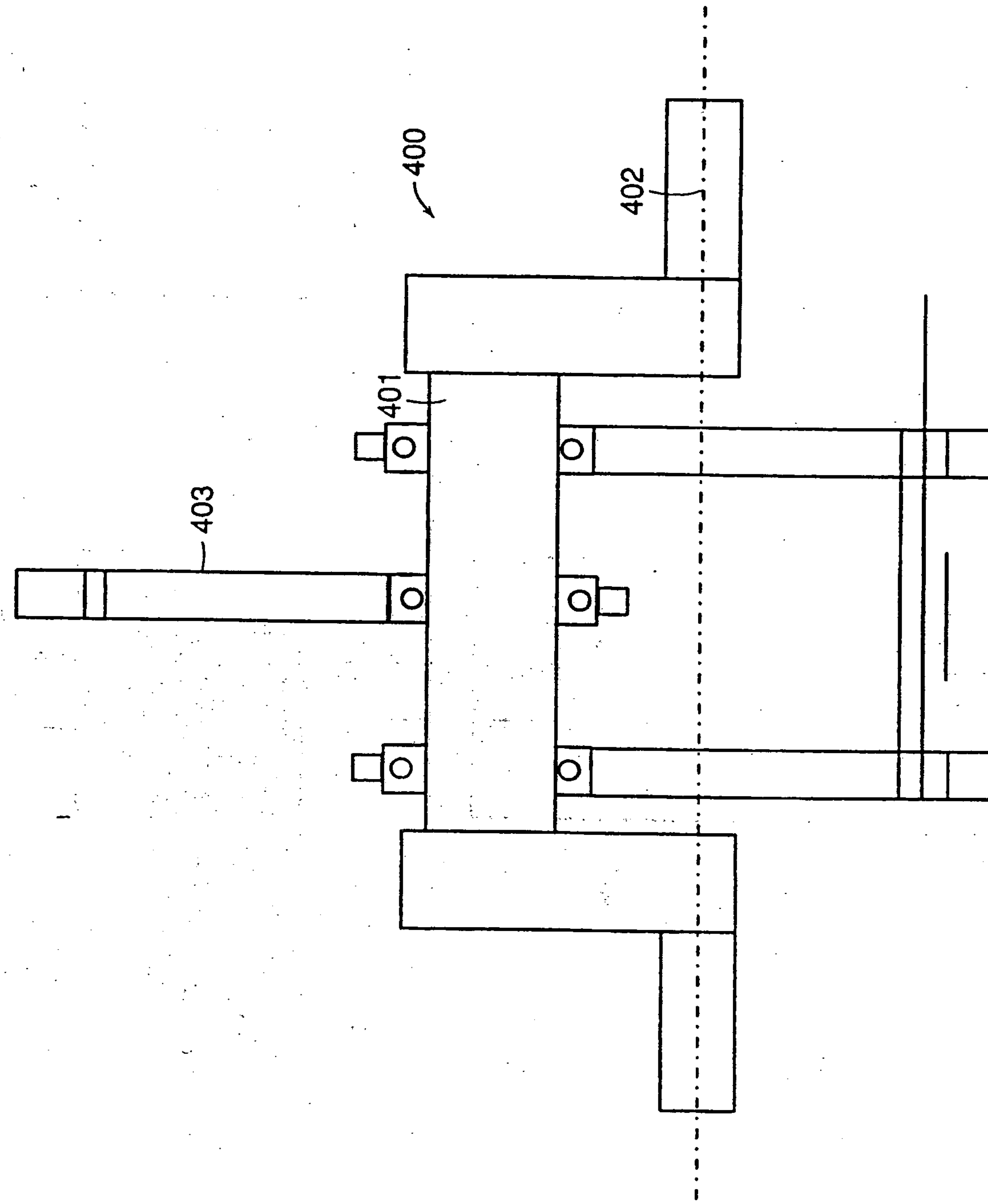

FIG. 7

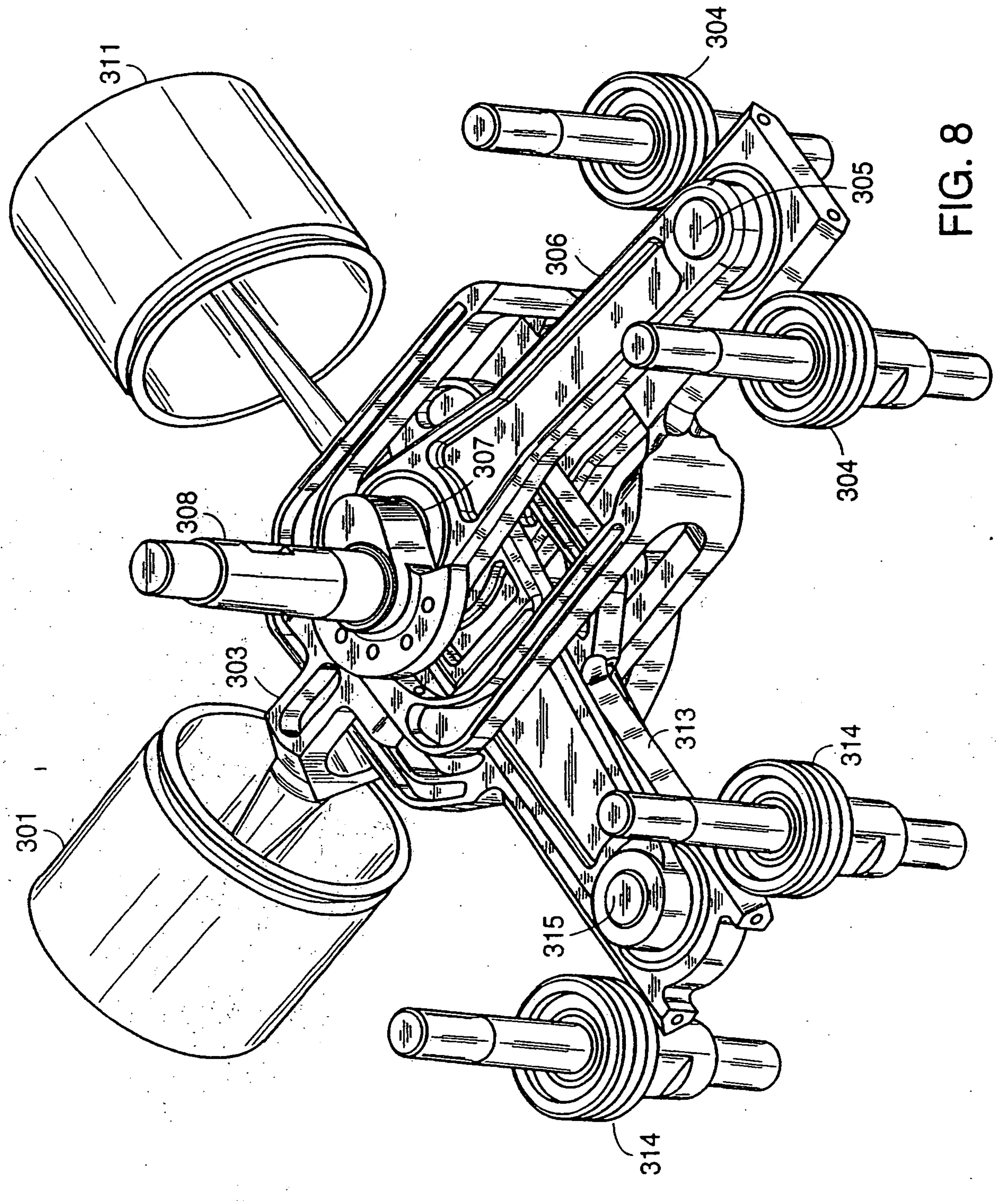

FIG. 8

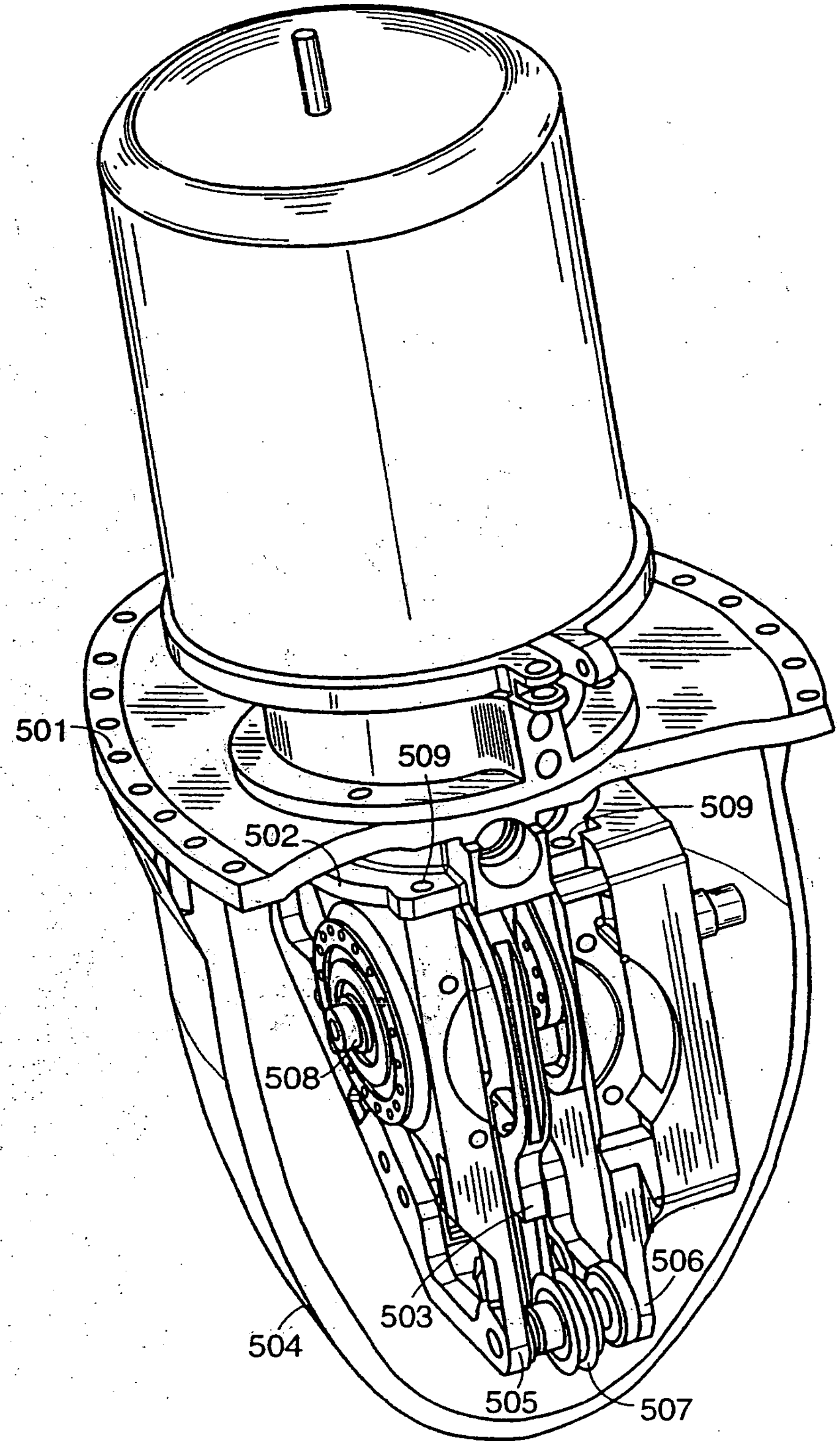

FIG. 9a

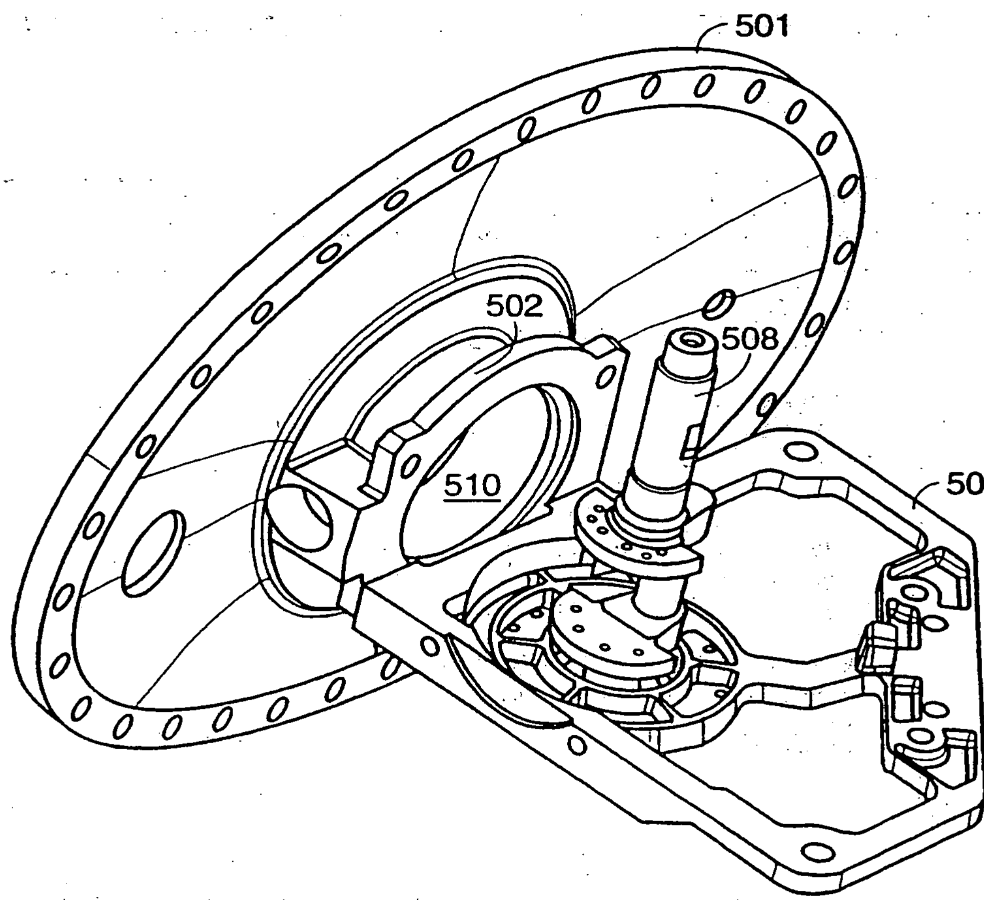

FIG. 9b

INTERNATIONAL SEARCH REPORT

International Application No
PCT/US 00/01931

A. CLASSIFICATION OF SUBJECT MATTER
IPC 7 F02G1/044 F01B9/02 F02B75/32

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 7 F02G F01B F02B

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category *	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	FR 2 721 982 A (LAKHDAR CHAKER) 5 January 1996 (1996-01-05)	1-4,6-9, 20,21, 23,24
Y	figures 1,9 abstract page 6, line 1 - line 24	10-19
X	US 1 769 375 A (LEARY) 1 July 1930 (1930-07-01) figure 1 page 2, line 1 - line 43	1-4,6-9
A	FR 2 067 119 A (GUILLON MARCEL) 20 August 1971 (1971-08-20) figure 1 claims 1-4	1-4,6-9

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[X] Further documents are listed in the continuation of box C.

[X] Patent family members are listed in annex.

* Special categories of cited documents :

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"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
19 April 2000	27/04/2000

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL – 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+31-70) 340-3016	Wassenaar, G

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Form PCT/ISA/210 (second sheet) (July 1992)

International Application No
PCT/US 00/01931

C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	DE 40 18 943 A (KRAUCH HELMUT PROF DR RER NAT ;MAURER THOMAS DIPL ING (DE)) 19 December 1991 (1991-12-19)	10-19
A	figure 1 abstract figure 2 claims 1-5	1-4
A	DE 42 05 283 A (BAYERISCHE MOTOREN WERKE AG) 26 August 1993 (1993-08-26) figure 1 abstract	4-6,22, 24
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PCT/US 00/01931

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US 1769375	A	01-07-1930	NONE		
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DE 4205283	A	26-08-1993	NONE		
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